CONCEPT OF TRANSDUCER BASE FOR HUMS OF SUSPENSION SYSTEM OF HIGH MOBILITY WHEELED VEHICLE

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Abstract: The paper introduces the concept of a transducer base for HUMS of the suspension system of a high mobility wheeled vehicle. The basic principles of the maintenance system with HUMS as a subsystem and its structure are clarified. The main benefits for operators of the maintenance system after implementation of HUMS into vehicles are presented. Preliminary ranges of loads to be monitored, generated in a selected point in the suspension system of a vehicle with a GVW of 3.5T class are presented, as well as general dimensions and material selection for the base of the transducer.

Keywords: high mobility wheeled vehicle, durability, suspension, transducer, CBM, HUMS, maintenance system

1. INTRODUCTION

High-mobility wheeled vehicles are complex technical objects with advanced capabilities and require an appropriate maintenance system for effective use, in which the level of service provided is suitably high. The current level of technical development and new requirements result in vehicles being equipped with extensive systems for monitoring the operational performance of individual vehicle sub-systems. This can be seen mainly in relation to the engine and transmission, as well as steering and braking systems (On-Board Diagnostics - OBD). Important (critical) and challenging to diagnose is the spring element in the vehicle's suspension system. In the case of high mobility vehicles, it is essential for maintaining vehicle mobility and highly important for military vehicles.

Military high mobility wheeled vehicles are the basic means of transportation used in tactical tasks. They are used to transport soldiers, cargo and are a platform for installing weapons systems [1]. However, the number of vehicles in the structures of the army is limited and there are no vehicles surplus (the problem of shortage of military equipment in the Ukraine). From this it follows that any military vehicle that has an expected transport task to perform should stay ready for use throughout its service life. Thus, the decision to introduce a vehicle into the maintenance system must be preceded by appropriate studies to determine the vehicle's durability and reliability [2-5]. The vehicle's lifetime resource is utilized in training (peacetime) and in tactical operations (wartime). Maintaining vehicles in an operational environment in readiness for use when there is a high time pressure and limited access to technical facilities requires highly qualified technical personnel and an effective operation management system. A simple but ineffective maintenance system is to take action after a failure has occurred (Reactive Maintenance - RM). Another approach (Preventive Maintenance - PM) involves performing

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maintenance activities, the scope of which is determined in advance based on experience and the results from mileage and stand tests conducted in advance (on a representative sample of vehicles). These activities are linked to a selected performance indicator (e.g., mileage, operating time, etc.). However, this system (PM) is also inefficient. A much more effective maintenance system is one where the scope of required service activities is determined based on diagnostic tests performed as part of a vehicle condition assessment (Proactive Maintenance - PaM). However, the timing of the tests is set in advance, and the scope of service activities to be performed depends on the result of the completed diagnostics. Postponing the timing of the diagnostic test is possible, but this decision is influenced by the overall level of the vehicle's condition and the driver's training, who must be able to recognize the pre-failure states of the vehicle. In practice, combined operating systems are often used: diagnosing the vehicle's condition according to a set schedule and linking the scope of repairs to the result of the diagnosis (e.g., passenger cars), and in the case of systems that are difficult to diagnose, using a system of planned maintenance activities (e.g., MBT, AFV, APC). However, the increasing complexity of vehicles means that the system of maintenance for effective operation cannot rely only on the training of drivers and diagnosticians but should be based on objective indications derived from the true technical condition of the vehicle assessed on a continuous rather than discrete basis (Predictive Maintenance - PdM). The progression of core maintenance strategies is shown in Figure 1.

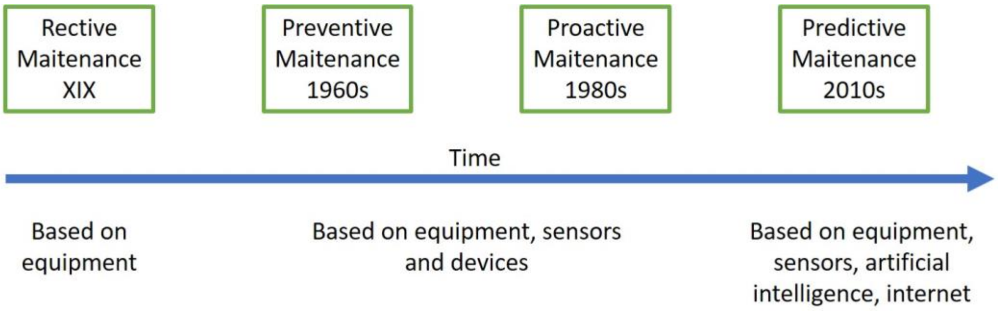
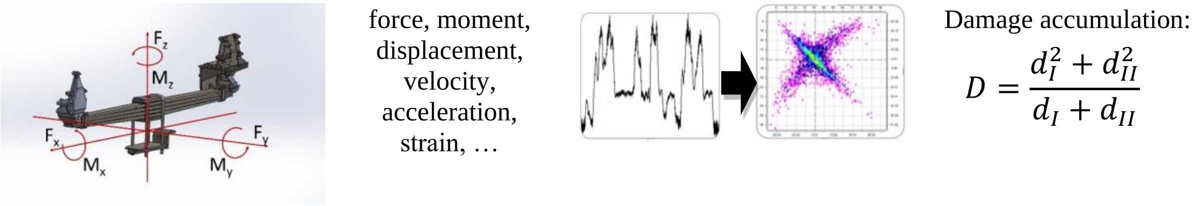


Fig. 1. The progression of core maintenance strategies.

However, the adaptation of vehicles for integration into such a system of operation (PdM) requires the compliance with several requirements that are essential and complex to implement: it is necessary to identify the vehicle systems for condition monitoring, determine the type of load impacting the degradation of the system and link its value to the selected measurement signal, design a monitoring system that records the values of the indicated measurement signal, and adopt a computational model using the recorded measurement signals. The functional diagram of such a measurement system is shown in Figure 2.



a) Mechanical system b) Loads c) Data transformation d) Result

Fig. 2. Idea of HUMS: a) place of installation – e.g. suspension system; b) physical quantities measured by the HUMS; c) data transformation, e.g. from time series into rainflow matrix; d) mathematical model of the analyzed phenomenon.

The acquired data can then be processed online or offline. Based on the processed data, the technical condition of the monitored vehicle system can be evaluated [4], such as assessing the probability of failure and its possible effects (Failure Mode and Effect Analysis - FMEA [5]). Finally, a system is needed to support operational decision-making [6, 7]. The implementation of the functions outlined is realistically possible by implementing a Reliability Centered Maintenance - RCM system coupled with Condition Based Maintenance - CBM, which is integrated with Health and Usage Monitoring System - HUMS [8].

2. RELIABILITY CENTERED MAINTENANCE – RCM

Reliability is a set of properties of a vehicle that describe its readiness for use as the probability of maintaining a state of readiness for use (operation ability) over an assumed period of time and under assumed operating conditions [9]. Reliability Centered Maintenance is a system that can be used at the tactical and operational levels to support decision-making, indicate the optimal maintenance actions for each vehicle and maintain the required operational, economic and safety indicators [10]. The place of RCM in relation to common maintenance strategies is shown in Figure 3.

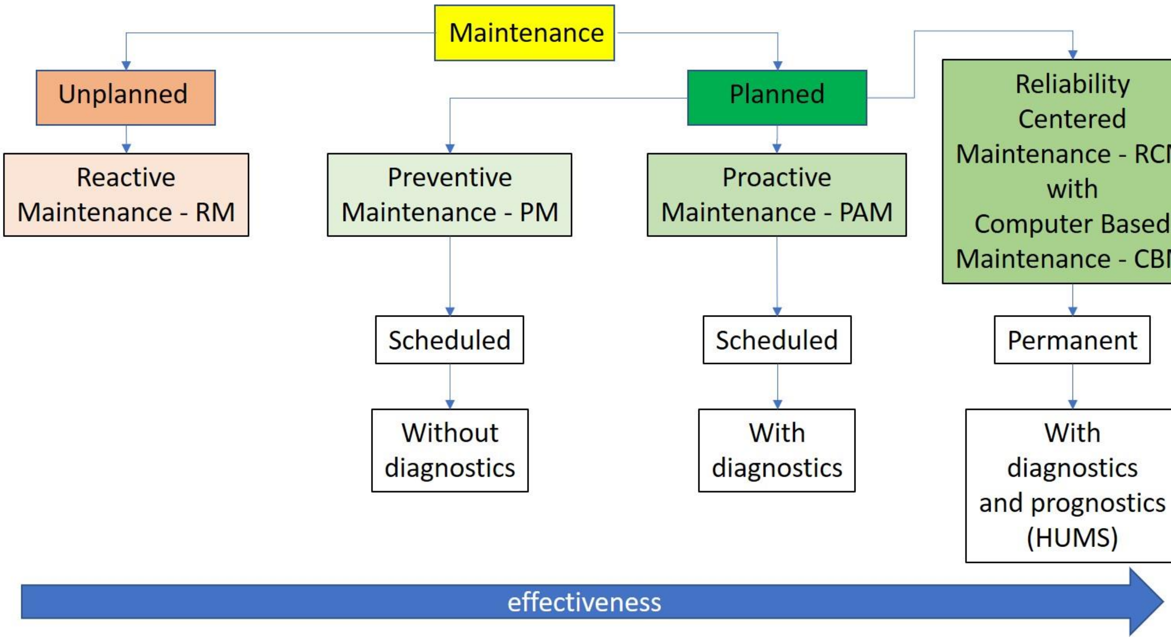


Fig. 3. Common types of maintenance strategies [10].

The Reliability Centered Maintenance strategy assumes that the technical condition of the components of a complex technical system (vehicle) has different importance to the tasks performed. It is also assumed that the same components in different vehicles of the same type are in different technical conditions (are worn out to different degrees) and, therefore, the risk of breakdown and non-completion of the task is different. The RCM model also considers the availability of technical facilities, people performing maintenance activities, or existing restrictions on access to parts and materials. By using RCM, one can reduce the risk of the consequences of undetected failures, the exposure of users to the dangers of vehicle malfunction, the need to assist an at-risk crew, and the reduction of users' confidence in their vehicles. Ultimately, a list of the most reliable vehicles can be compiled and matched to planned transportation tasks that have different priorities for being completed [11-13]. This system (RCM) uses data developed by a subsystem - Condition Based Maintenance - CBM.

3. CONDITION BASED MAINTENANCE – CBM

The Condition-Based Maintenance strategy is based on a response that is tailored to the current needs of the vehicle (technical system). In this strategy, service activities that are not necessary to be performed at the moment are avoided, and an organizational effort is focused only on necessary assignment. However, it is required that the components of the technical system (vehicle) are undergoing changes in technical condition long enough in time so that it is possible to recognize the technical state and arrange service actions before the failure occurs.

The main benefits of implementing a CBM system are listed below:
- service actions are performed before the borderline condition of the monitored vehicle system is reached;
- the cost of the troubleshooting activities is minimized;
- the overall level of reliability of the technical system is improved;
- vehicle downtime is reduced;
- actions are taken for which suitable preparation can be made in advance;

- the system allows individual vehicle maintenance activities to be carried out in an optimized manner;
- very high operational awareness with respect to each vehicle;
- minimizing the risk of not accomplishing the mission due to vehicle (technical system) failures - this applies to operational failures.

The most important limitations in the implementation of the CBM system:
- difficulty in indicating easily measurable and limited number of diagnostic signals providing sufficient data for planning maintenance activities;
- a reliable and durable monitoring system for the operation of the monitored vehicle systems (technical system) is required, which is costly and complicated;
- the required level of training the people involved in service actions is high, which demands support measures (access to the knowledge base, training);
- some forms of wear (such as fatigue) are difficult to monitor;
- the timing of service actions is not known in advance, which makes it difficult to optimally organize the service structure.

The main differences between CBM and PAM are shown in Table 1.

Table 1. Characteristic of CBM and PAM

CBM	PAM
The system is based on continuous analysis of diagnostic signals to look for symptoms concerning changes in the technical condition of the monitored component.	The system is based on periodically analyzing diagnostic signals to look for symptoms concerning changes in the technical condition of the monitored component.
Decisions developed by the system are uninfluenced by the subjective assessment of the diagnostician and the driver.	Decisions are made considering the subjective assessment of the diagnostician and the driver.
The accuracy of decisions depends on the precision of decision-making algorithms and the quality of analysis of selected signals.	The accuracy of the decision depends on the training of drivers and diagnosticians and the available diagnostic equipment.
Optimized costs in relation to the results obtained (vehicle reliability, reducing the risk of not completing the task).	Predicting how long a vehicle will operate correctly is subject to risk due to decision-making by the diagnostician and the driver.
Multi-stage information about significant changes in the technical condition of the monitored component, giving time to make the optimal decision.	Acquire diagnostic data only after tests are performed periodically.

Reviewing the summary of characteristics in Table 1, it can be seen that the data in the CBM system is more up-to-date and objective, the decisions made can be made more quickly and accurately, and the time to take action is extended, which is very important with regard to military vehicles. Examples of the application of RCM strategies can be found in [14, 15]. However, it is crucial to ensure continuous access to reliable data relating to the technical condition of the monitored component, preferably already preprocessed. These and other requirements were the basis for the development of systems for monitoring the health and usage of technical objects – HUMS [16-18].

4. IDEA OF HEALTH AND USAGE MONITORING SYSTEM - HUMS

The primary task of the HUMS system is to monitor changes in the technical condition of the observed object, along with recording selected loads on this element and to describe the impact of these loads on the current and/or future technical condition of the element unambiguously. The determination of a trend line describing the change in the technical condition of an element, together with the determination of a limiting condition, makes it possible to inform the operating system in advance of an increased risk of damage to the element and a possible change in the technical condition of the vehicle. The basic functional model of the HUMS system is shown in Figure 4.

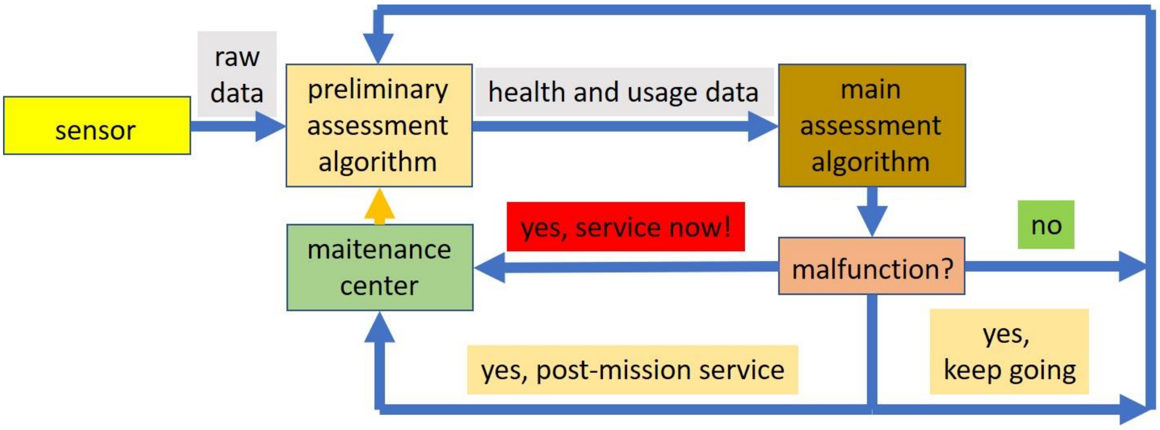


Fig. 4. Diagram of HUMS.

The HUMS system consists of an alignment of sensors that record the operating parameters of the observed component (subassembly) and computational procedures that allow continuous comparison of operating parameters with reference data. The main problem and task are to build as simple a monitoring system as possible (minimum number of sensors), while offering a set of required and reliable data. Several armies around the world are researching the development and implementation of an effective HUMS system for military vehicles [8, 19, 20].

5. SENSOR FOR MEASUREMENT OF FORCES AND MOMENTS (6DOF) IN SUSPENSION SYSTEM

Concerning the suspension systems of high-mobility wheeled vehicles, a spring component (e.g., spring, coil spring, torsion bar) can be considered a critical component whose failure provides a real threat of not completing the mission. If it is destroyed/damaged, the vehicle will lose its mobility and require emergency service measures, often under hostile conditions.

In the basic classical suspension system of an off-road vehicle, the spring is responsible for reducing the loads from the road, acting on the sprung mass. It also provides for the relative displacement of the wheels and guidance of the vehicle axles [21]. It is subjected to complex loads, the course of which affects its durability [22]. The load sequence, which is influenced by, among other things, vehicle design, operating conditions, or the driver's driving style [23], determines the durability and reliability of the vehicle, and indirectly the operational safety of the entire organization.

To directly measure the values of these loads, (in this case to measure forces and moments), a measuring system equipped with a suitable transducer is required. Road and laboratory tests use off-the-shelf measurement kits equipped with wheel load transducers (wheel force sensors) [24-26], installed to the wheel rim. Due to their design and installation position, as well as the resulting expense of purchase, they are not used as a permanent component of the vehicle's suspension system and cannot be a permanent source of data for the vehicle's HUMS system. This creates an area to look for other solutions that have greater potential for series applications.

5.1. Estimation of loads acting on the transducer base

To design a vehicle-dedicated prototype of a suitable transducer, it is necessary to determine the type of road loads to be measured and the values of these loads, as well as a suitable installation location. No less important is the design of the measurement system for implementation on the transducer's support structure.

It was assumed that the designed transducer would be used to measure the forces and moments acting on the spring in the suspension system of the rear axle of a lightweight high mobility vehicle. A diagram of the acting forces and moments is shown in Figure 5.

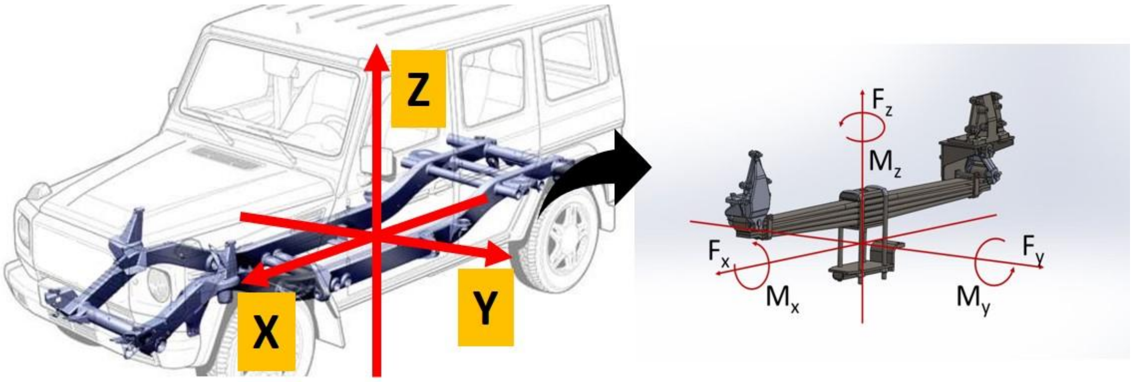


Fig. 5. Active forces and moments in suspension system.

Examples of vehicles in the analyzed class could be Jeep Wrangler II [27], Suzuki Jimny [28], Kia Retona [29], UAZ 469B [30]. A summary of selected technical data of the listed vehicles is shown in Table 2. When estimating the load distribution, a 60:40 split was assumed for the vehicle without a load and 40:60 with a load.

Table 2. Load distribution in selected vehicles.

	Jeep Wrangler	Suzuki Jimny	Kia Retona	UAZ 469B
Running gear	4x4	4x4	4x4	4x4
Kerb weight [kg]	1500	1200	1510	1540
Front axle (60%)[kg]	900	720	900	920
Rear axle (40%) [kg]	600	480	610	620
GVW [kg]	2100	1530	1900	2290
Front axle (60%)[kg]	840	610	760	960
Rear axle (40%) [kg]	1260	920	1140	1330

For further calculations (due to reduce the cost of future research), the loads were assumed to a typical UAZ 469B (3,5T class) vehicle ($m = 1700\ [kg]$). Considering that: the mass of the rear axle with wheels and half the mass of the drive shaft is $150\ [kg]$ and the car will be 50% loaded, then a static vertical force will act on a single rear spring $F_Z = 4000\ [N]$.

To estimate the effect of inertia forces on the load on the spring, it was assumed that the COG is located in the central plane of the vehicle, at the height of the $H = 1000\ [mm]$ between the front and rear axles. Assuming that the wheelbase of the vehicle is $WB = 2300\ [mm]$, the track width is $TW = 1442\ [mm]$, the longitudinal ratio i_Y in the suspension system is:

$$i_Y = \frac{H}{WB} = \frac{1000}{2380} = 0{,}42 \tag{1}$$

and the transverse ratio i_X:

$$i_X = \frac{H}{TW} = \frac{1000}{1442} = 0{,}69 \tag{2}$$

This means that in the case of braking, for example: $a = 5\ m/s^2$(maximum deceleration for the selected vehicle [30]), longitudinal inertia force F_{Xd}:

$$F_{Xd} = ma = 1700 \cdot 5 = 8500\ [N] \tag{3}$$

operating with the i_Y ratio results in an increase in the load on the front axle and a decrease in the load on the rear axle by ΔF_{ZXd}:

$$\Delta F_{ZXd} = F_{Xd} \cdot i_Y = 8500 \cdot 0{,}42 = 3570\ [N] \tag{4}$$

This means that a reduced force is applied to a single rear spring during braking F_{ZXd}:

$$F_{ZXd} = F_Z - \Delta F_{ZXd} = 4000 - 0{,}5 \cdot 3570 = 2215\ [N] \tag{5}$$

When accelerating a vehicle with acceleration, for example.: $a = 2\ m/s^2$ (maximum acceleration for the selected vehicle [30]), the force of inertia F_{Xa} :

$$F_{Xa} = 1700 \cdot 2 = 3400\ [N] \tag{6}$$

operating with the i_Y ratio results in an increase in the load on the rear axle and a decrease in the load on the front axle by ΔF_{ZXa}:

$$\Delta F_{ZXa} = F_{Xa} \cdot i_Y = 3400 \cdot 0{,}42 = 1428\ [N] \tag{7}$$

This means that there is an increased force on the single rear spring during acceleration F_{ZXa}:

$$F_{ZXa} = F_Z + \Delta F_{ZXa} = 4000 + 0{,}5 \cdot 1428 = 4714\ [N] \tag{8}$$

During cornering, there is a lateral inertia force F_Y applying load to the outer springs and relieving load on the inner springs. Assuming that the coefficient of traction of the wheels to the road $\mu = 0{,}7$, then the change in vertical force in curved motion ΔF_{ZY} can amount :

$$\Delta F_{ZY} = F_Z \cdot \mu \cdot i_X = 4000 \cdot 0{,}7 \cdot 0{,}69 = 1932\ [N] \tag{9}$$

This means that the rear outer spring, when driving on a curve of the road, can be loaded with force F_{ZYo}:

$$F_{ZYo} = F_Z + \Delta F_{ZY} = 4000 + 1932 = 5932\ [N] \tag{10}$$

and a rear inner spring F_{ZYi}:

$$F_{ZYi} = F_Z - \Delta F_{ZY} = 4000 - 1932 = 2068\ [N] \tag{11}$$

Combining the two cases (braking/acceleration and cornering), the rear spring can be loaded by the vertical force F_{ZS} , the values of which are shown in Table 3.

Table 3. Values of forces to different conditions of vehicle movement.

	braking	braking and turning	acceleration	acceleration and turning
outer spring	$F_{ZXd} = 3570\ [N]$	$F_{ZXYdo} = 5682\ [N]$	$F_{ZXa} = 4714\ [N]$	$F_{ZXYdo} = 6646\ [N]$
inner spring	$F_{ZXd} = 3570\ [N]$	$F_{ZXYdi} = 1638\ [N]$	$F_{ZXa} = 4714\ [N]$	$F_{ZXYdi} = 2782\ [N]$

Thus, the range of load changes in the spring is:
- in the X axis: $\Delta F_X = 3400\ \div 8500\ [N]$;
- in the Y axis: $\Delta F_Y = 1352\ \div 4152\ [N]$;
- in the Z axis: $\Delta F_Z = 1638\ \div 6646\ [N]$.

In addition to the loads from the forces F_X, F_Y, F_Z the springs are also affected by the moments of forces M_X, M_Y, M_Z from braking/acceleration, body tilt and angular displacement of the vehicle axle during off-road and cross-country driving, respectively. Assuming that the dynamic radius of the wheel $R = 0{,}4\ [m]$, the maximum braking moment M_{Yd}:

$$M_{Yd} = F_{ZXd} \cdot \mu \cdot R = 2215 \cdot 0{,}7 \cdot 0{,}4 = 620\ [Nm] \tag{12}$$

and the maximum acceleration moment M_{Ya}:

$$M_{Ya} = F_{ZXa} \cdot \mu \cdot R = 4714 \cdot 0{,}7 \cdot 0{,}4 = 1320\ [Nm] \tag{13}$$

Similarly, the values of the tilting moment M_X can be determined:

$$M_X = F_Z \cdot \mu \cdot (H - R) = 4000 \cdot 0{,}7 \cdot (1 - 0{,}4) = 1670\ [Nm] \tag{14}$$

In the case of the moment M_Z its value can be estimated by bringing the conditions of motion to the case when one wheel encounters a bump, which leads to deflection of the spring (straightening). Assuming that the driving force is then transmitted through one wheel, the moment M_Z:

$$M_Z = F_{ZXYdo} \cdot \mu \cdot B = 6646 \cdot 0{,}7 \cdot 1{,}442 = 6708\ [Nm] \tag{15}$$

However, it should be noted that the action of the moment M_Z is limited by the allowable angular displacement of the rear axle in the horizontal plane, which is equal to the difference in the distance of the axle from the front attachment point of the spring to the frame when it is loaded and unloaded. In practice, this value is small (a few mm) and the moment M_Z is transmitted to the spring to a limited extent. It is tentatively assumed that this value is $6700\ [Nm]$.

Summary of estimated moment values:
- $M_X = \pm 1670\ [Nm]$;
- $M_Y = \pm 1320\ [Nm]$;
- $M_Z = \pm 6700\ [Nm]$.

5.2. Geometry and material of the transducer base

The base of the transducer with a suitable design, made of a material that is supposed to be able to handle the estimated forces and moments, should deform in elastic range to enable the accurate measure of the load by the measurement system. At the same time, the real loads should not damage the transducer. Hence, the material often selected for the transducer base is steel, such as: 1.8159 (51CrV4), 1.4548 (X5CrNiCuNb17-4-4), copper, e.g.: 2.1247 (CuBr2) or aluminum, e.g.: 3.1355 (AlCuMg2) [31]. The appropriate material selection must be preceded by a stress analysis to the pre-assumed design of the transducer base. The calculation flowchart is shown as an example of the transducer concept, which was composed of two flat bars connected by a cylindrical deformable element, as simplified shown in Figure 6.

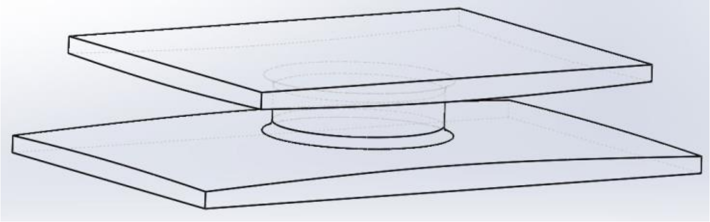


Fig. 6. Sketch of the transducer base.

For the presented sketch of the transducer base, the load model shown in Figure 7 was built.

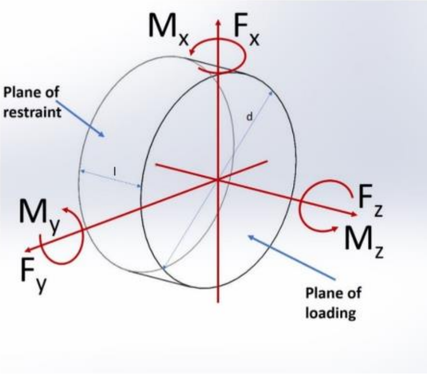


Fig. 7. Load model of the transducer base.

A summary of loads in the form of forces and moments for preliminary strength calculations was collected in Table 4.

Table. 4. A summary of loads.

F_x	F_y	F_z	M_x	M_y	M_z	d	l
8500 [N]	4152 [N]	6646 [N]	1670 [Nm]	1320 [Nm]	6700 [Nm]	55 [mm]	15 [mm]

Based on the theory of strength of materials, the influence of loads on the stresses induced was estimated according to the calculation flowchart presented:

- modulus of section bending strength W_g:

$$W_g = \frac{\pi d^3}{32} = \frac{\pi \cdot 55^3}{32} = 16334\ [mm^3] \tag{16}$$

- modulus of section torsion strength W_s:

$$W_s = \frac{\pi d^3}{16} = \frac{\pi \cdot 55^3}{16} = 32668\ [mm^3] \tag{17}$$

- cross-sectional area A:

$$A = \frac{\pi d^2}{4} = \frac{\pi \cdot 55^2}{4} = 2376\ [mm^2] \tag{18}$$

- normal stress σ_{Z1}:

$$\sigma_{Z1} = \frac{F_Z}{A} = \frac{6646}{2376} = 2{,}8\ [MPa] \tag{19}$$

- normal stress σ_{Z2}:

$$\sigma_{Z2} = \frac{F_X \cdot l + M_Y}{W_g} = \frac{8500 \cdot 15 + 1320 \cdot 10^3}{16334} = 88{,}6\ [MPa] \tag{20}$$

- normal stress σ_{Z3}:

$$\sigma_{Z3} = \frac{F_Y \cdot l + M_X}{W_g} = \frac{4152 \cdot 15 + 1670 \cdot 10^3}{16334} = 106{,}1\ [MPa] \tag{21}$$

- shear stress τ_{XY1}:

$$\tau_{XY1} = \frac{F_X}{A} = \frac{8500}{2376} = 3{,}6\ [MPa] \tag{22}$$

- shear stress τ_{XY1max}:

$$\tau_{XY1max} = \frac{16F_X}{3\pi d^2} = \frac{16 \cdot 8500}{3 \cdot \pi \cdot 55^2} = 4{,}8\ [MPa] \tag{23}$$

- shear stress τ_{XY2}:

$$\tau_{XY2} = \frac{F_Y}{A} = \frac{4152}{2376} = 1{,}7\ [MPa] \tag{24}$$

- shear stress τ_{XY2max}:

$$\tau_{XY2max} = \frac{16F_Y}{3\pi d^2} = \frac{16 \cdot 4152}{3 \cdot \pi \cdot 55^2} = 2{,}3\ [MPa] \tag{25}$$

- shear stress τ_{M_Z}:

$$\tau_{M_Z} = \frac{M_Z}{W_S} = \frac{6700 \cdot 10^3}{32668} = 205\ MPa \tag{26}$$

A summary of the stresses was shown in Table 5.

Table 5. A summary of the stresses.

σ_{Z1}	σ_{Z2}	σ_{Z3}	τ_{XY1}	τ_{XY2}	τ_{M_Z}
$2{,}8\ [MPa]$	$88{,}6\ [MPa]$	$106{,}1\ [MPa]$	$3{,}6\ [MPa]$	$1{,}7\ [MPa]$	$205\ MPa$

Reduced stress according to the Huber-Mises-Hencky (HMH) hypothesis:

$$\sigma_{zr}=\frac{1}{\sqrt{2}}\sqrt{(\sigma_X-\sigma_Y)^2+(\sigma_Y-\sigma_Z)^2+(\sigma_Z-\sigma_X)^2+3({\tau_{XY}}^2+{\tau_{YZ}}^2+{\tau_{XZ}}^2)} \tag{27}$$

- sum of normal stresses $\sum\sigma$:

$$\sum\sigma=\sigma_{Z1}+\sigma_{Z2}+\sigma_{Z3}=2{,}8+88{,}6+106{,}1=197{,}5\ [MPa] \tag{28}$$

- sum of shear stresses $\sum\tau$:

$$\sum\tau=\tau_{XY1}+\tau_{XY2}+\tau_{M_Z}=3{,}6+1{,}7+205=210{,}3\ [MPa] \tag{29}$$

By substituting $\sum\sigma$ and $\sum\tau$ into equation (26), we get:

$$\sigma_{zr}=\frac{1}{\sqrt{2}}\sqrt{(0-0)^2+(0-197{,}5)^2+(197{,}5-0)^2+3(210{,}3^2+0^2+0^2)}=324{,}6\ [MPa] \tag{30}$$

With the estimated value of shear and normal stresses, as well as reduced stress, it is possible to select the material for the base of the transducer. It was assumed initially that it could be a steel e.g: 1.8159 (51CrV4). The chemical composition of steel 1.8159 was presented in Table 6.

Table 6. Chemical composition of steel 1.8159 [32].

Steel designation		C [%]	Si (max) [%]	Mn [%]	P (max) [%]	S [%]	Cr [%]	V [%]
51CrV4	1.8159	0,47-0,55	0,40	0,70-1,10	0,025	0,025	0,90-1,20	0,10-0,25

The steel 1.8159, after quenching and tempering, can achieve the strength parameters shown in Table 7.

Table 7. Strength properties of steel 1.8159 after quenching and tempering ($d=55\ [mm], l=15\ [mm]$) [32].

$R_m[MPa]$	$R_e[MPa]$	$A[\%]$	$Z[\%]$	$KV[J]$
~ 1100	>800	≥ 12	≥ 50	≥ 30

Parameters of hardness of 1.8159 steel was presented in the Figure 8.

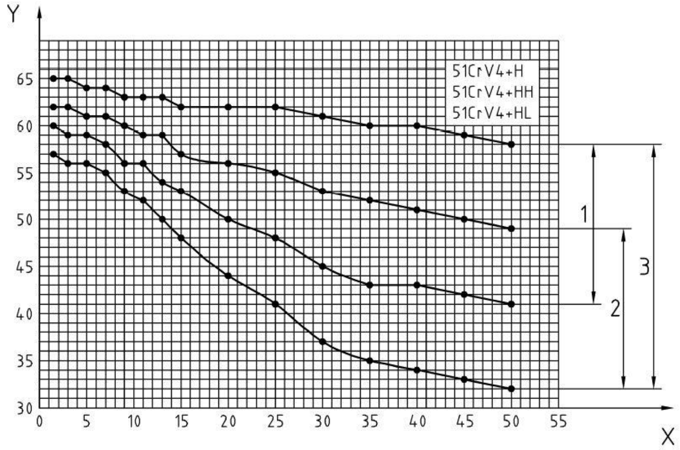


Fig. 8. Scater bands for the Rockwell – C hardness in the end quench hardenability test [32]
X – distance from quenched end of test piece, mm; Y – hardness, HRC; HH – grade; HL-grade; H-grade.

An appropriate arrangement of strain gauges, elements that change their resistance depending on the strain, placed on the base of the transducer, makes it possible to indirectly record the course of the identified loads. The article does not provide details of the arrangement of strain gauges on the transducer base due to the required scope of the description, which will be the subject of a separate paper.

6. SUMMARY AND CONCLUSIONS

The analyses presented in the article were intended to point out the benefits of implementing a HUMS system into a maintenance system, which can then be reliability-based (RCM), providing the opportunity to individualize maintenance tasks, reduce maintenance costs and ensure the required technical readiness and reliability of vehicles. In the HUMS system, as shown in Figure 4, two components are key: a sensor (transducer) and a computational module that uses the recorded loads to assess the technical condition of the monitored component. The location of the transducer and the derived possible types of loads to be recorded are indicated. The values of such loads were estimated based on the calculations presented. In the next step, computations were presented, based on which the resulting stresses in the base of the transducer were determined. On this basis, it was possible to propose a material for making the transducer. Having the transducer base done, it will be possible to complete it with elements sensitive to the resulting deformations (strain gauges), and on this basis, after scaling the entire measurement system, it will be possible to identify loads.

To carry out an automated assessment of the monitored component, it will be necessary to develop a computational model relating to the multiaxial fatigue course.

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